

Detailed reports are also provided at regular intervals to explain any corrective actions taken and any pending actions. In accordance with governmental regulations, any changes made to the machinery management system are documented along with alarm setpoints, calibration information, and serial numbers.

Having Bently Nevada perform the

instrumentation maintenance as part of a site management program reduces your costs and increases the dependability of your machinery management system. Bently Nevada's Service organizations will perform any necessary maintenance and advise your personnel of the actions taken, in writing.

Our Machinery Diagnostic Services

organization is part of a complete Site Management Agreement. Facilities with machinery management systems that are equipped with remote telecommunications can contact a qualified engineer, 24 hours a day, for advice on possible machinery problems. Upcoming Orbit articles will discuss our products' remote machinery diagnostic capabilities. ■

Continued from page 17 (Some realities of field balancing).

Dynamic Stiffness

Dynamic stiffness is the restraint of motion when a mechanical system is subjected to an oscillating or rotating force. This can be expressed in the general equation **Motion = Force/Restraint**. Dynamic stiffness is the sum of the stiffnesses associated with each of the physical characteristics of a system: spring, damping, and mass. Each of these stiffnesses results in a characteristic lag in the response of the system with respect to the applied force. This is most easily seen in the classical, one-dimensional spring-damper-mass model, (Figure 1).

In the rotating, two-dimensional rotor/bearing/seal system (Figure 2) each stiffness component, including that of the surrounding fluid, results in a distinct **direction** of response. The direction is associated with the phase lags seen in the one-dimensional case. When the system is unbalanced to provide an input force, the resulting motion is shown in Figure 2.

At low speeds, when the spring stiffness is dominant, the motion is in the direction of the unbalance force F (push on a spring and it moves in the direction of the force). It is

determined by the ratio $\frac{F}{K}$, where K is the spring stiffness.

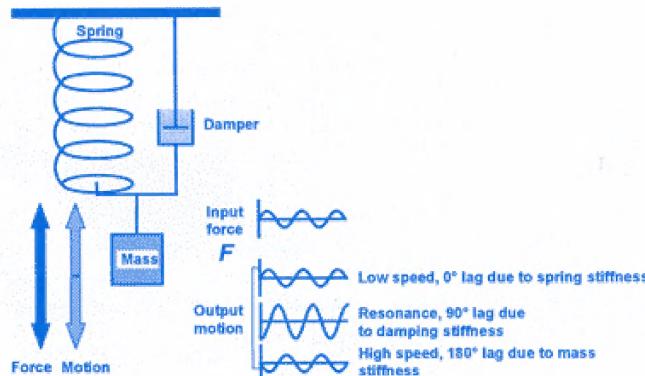


Figure 1
Response of a one-dimensional spring-damper-mass system to an oscillating force.

At resonance, when the spring and mass stiffnesses cancel, the motion lags the force by 90° (against the direction of rotation). The motion is in quadrature to the force and is determined by the ratio $\frac{F}{D(1-\lambda)\Omega}$, where $D(1-\lambda)\Omega$ † is the fluid wedge (tangential or cross-coupled) portion of the dynamic stiffness (push on a wedge and it moves at right angles to the force).

At high speeds, when mass stiffness is dominant, the motion lags the force by 180° (a track and field athlete swinging a hammer uses his mass to balance the system by moving away from the weight). The motion is determined by $\frac{F}{M\Omega^2}$, where $M\Omega^2$ is the mass portion of the dynamic stiffness.

The general expression for synchronous motion, then, is

$$\text{Motion} = \frac{\text{Force}}{\text{Restraint}} = \frac{\text{Force}}{(K + D(1-\lambda)\Omega - M\Omega^2)}.$$

† The general Quadrature Stiffness term is $K_Q = jD(\omega - \lambda\Omega)$, where ω is rotor precession rate and λ is the Fluid Circumferential Average Velocity Ratio due to the fluid dragged into rotation by the shaft, which reduces the effect of damping. In this case, $\omega = \Omega$, so that

$$K_{Q \text{ Synch}} = D(1-\lambda)\Omega.$$

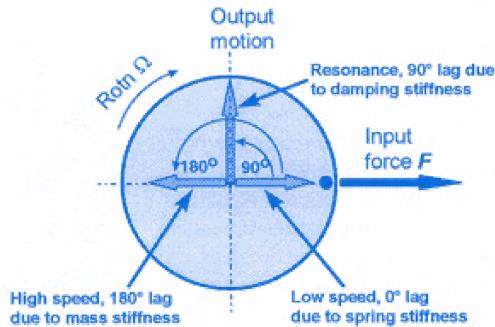


Figure 2
Response of a two-dimensional, rotating spring-damper-mass system to a force rotating Y to X (cw) at the same rate.